PROCEEDINGS

of the

15th MINI CONFERENCE ON VEHICLE SYSTEM DYNAMICS, IDENTIFICATION AND ANOMALIES

Held at the

Faculty of Transportation Engineering and Vehicle Engineering
Budapest University of Technology and Economics, Hungary
BUDAPEST, 7-9 November, 2016

Edited by Prof. I. Zobory

VSDIA 2016

BUDAPEST UNIVERSITY OF TECHNOLOGY AND ECONOMICS

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CONTENTS

CONTROL-ORIENTED MODELLING OF THE VARIABLE-GEOMETRY SUSPENSION FOR INDEPENDENT STEERING PURPOSES

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ABSTRACT

The paper presents the modelling and control design of an independent steering system which is based on the variable-geometry suspension. Through the actuation of the suspension the camber angle of the wheel is modified, which results in the variation of the steering wheel scrub radius. In the paper the model of the independent steering mechanism and the relationship between the steering and the suspension geometry are formulated. Furthermore, the model is validated through a high-fidelity multi-body suspension model Matlab/SimMechanics. In the paper the steps of the robust control design are presented, and the efficiency of the system is illustrated through different vehicle dynamical scenarios.

Keywords: variable-geometry suspension, independent steering, control-oriented modelling

1. INTRODUCTION AND MOTIVATION

The electrification of the vehicle drivelines provides a new possibility to enhance the stability and safety of road vehicles. A novel solution of the electric drive is the application of in-wheel electric motors. It makes possible the distribution of the traction forces between the wheels, by which additional functionalities can be achieved, e.g. the torque vectoring of the vehicle. The in-wheel electric drive offers new challenges in the steering of the vehicle, such as independent steering. The goal of the independent steering concept is to improve the lateral dynamics of the vehicle using individually controlled wheels. The independent steering control for the rear wheels to modify the toe angle is presented by [5]. The analysis of the independent wheel steering system for heavy vehicles is found in [10]. An indirect power steering measure called differential drive torque assisted steering is proposed by [14]. A fault-tolerant control approach for a four-wheel independently actuated electric vehicle to handle fault scenarios is proposed by [4].

In this paper a new solution of independent steering, which is based on the variable-geometry suspension solution, is proposed. The aim of the suspension control is the modification of the geometry, which results in a change in the camber or the toe angle. A rear-suspension active toe control for the enhancement of driving stability is proposed by [3]. The active tilt control system, which assists the driver in balancing the vehicle and performs tilting in the bend, is an essential part of a narrow vehicle system, see [8]. These vehicles require the design of innovative active wheel tilt and steer control strategies in order to perform steering similarly to a car on straight roads but in bends they tilt as motorcycles, see [13]. The advantages of the variable-geometry suspension are the simple structure, low energy consumption and low cost compared to other mechanical solutions such as an active front wheel steering, see [2], [6]. In the paper the control design of an independent wheel steering system for the front wheels is proposed. The novelty of the paper is the application of the variable-geometry suspension in the steering solution. The contributions are the modelling and validation of the suspension system.

The paper is organized as follows. The modelling of the variable-geometry suspension system is presented in Section 2. The validation and the linearization of the nonlinear model are found in Section 3. Section 4 demonstrates the efficiency of the steering system. Finally, Section 5 summarizes the contribution of the paper.

2. MODELLING OF INDEPENDENT STEERING

In this section the modelling of independent steering is proposed. The variable-geometry suspension performs the modification of the wheel position and orientation. Thus, the wheel camber angle and the scrub radius of the suspension vary. In the following, the formulation of the lateral vehicle model with the consideration of the camber angle and scrub radius is presented.

The goal of the variable-geometry suspension is to perform the wheel camber angle and the scrub radius modification. The camber angle results in a lateral force on the tyre-ground contact. Since the longitudinal force has a rotatory effect on the wheel, the scrub radius of the wheel influences the steering dynamics of the wheel. Therefore, a lateral force results from the wheel steering through the scrub radius modification.

Since the variable-geometry suspension has one actuator in each wheel, it is necessary to find a suspension construction with which the lateral force generation of the camber and the scrub radius is in coordination. Thus, the forces from the wheel tilting and the steering from the scrub radius have the same effect on the vehicle dynamics. In the automotive industry, there are the two commonly used suspension types: Double wishbone and MacPherson. Double wishbone suspension can be manufactured with relatively large nominal scrub radius, while the MacPherson type usually has a small nominal scrub radius, close to zero. Since the variable-geometry suspension has to be able to realize a negative value as well as a positive value of the scrub radius, in the paper the McPherson construction is used. The actuator is incorporated in the suspension between the wheel hub and the wheel. It is able to generate an active torque around B to tilt the wheel. However, it also has a counter effect M_{act} on the hub. In the McPherson construction the suspension is able to rotate around the connection point A of the chassis. Moreover, the arm connects the hub D and the chassis C with joints, which are able to guarantee the rotation and the motion of the suspension. The scheme of the variable-geometry suspension is shown in Fig. 1. Several forces influence the motion of the suspension and the wheel. The force of the suspension compression and damping F_{susp} is formulated as

$$F_{susp} = s_{susp}\left(\frac{z_w + z_{w,0}}{\sin\varepsilon_1}\right) + d_{susp}\,\frac{\dot{z}_w}{\sin\varepsilon_1} \tag{1}$$

where s_{susp} and d_{susp} are the stiffness and damping coefficients, $z_{w,0}$ is the joint position, resulting from the static suspension compression.

The lateral force acting on the tyre is F_y. It is derived from the Magic Formula, see Pacejka [2004]. F_{tyre} is the force from the tyre compression, which has a direction to the wheel:

$$F_{tyre} = s_{tyre}\,\frac{\left(r_w\cos\gamma - l_{tyre}\sin\gamma - r_w - z_w\right) + z_{tyre,0}}{\cos\gamma} \tag{2}$$

where s_{tyre} is the tyre stiffness, r_w is the wheel radius and $z_{tyre,0}$ is the static compression of the tyre.

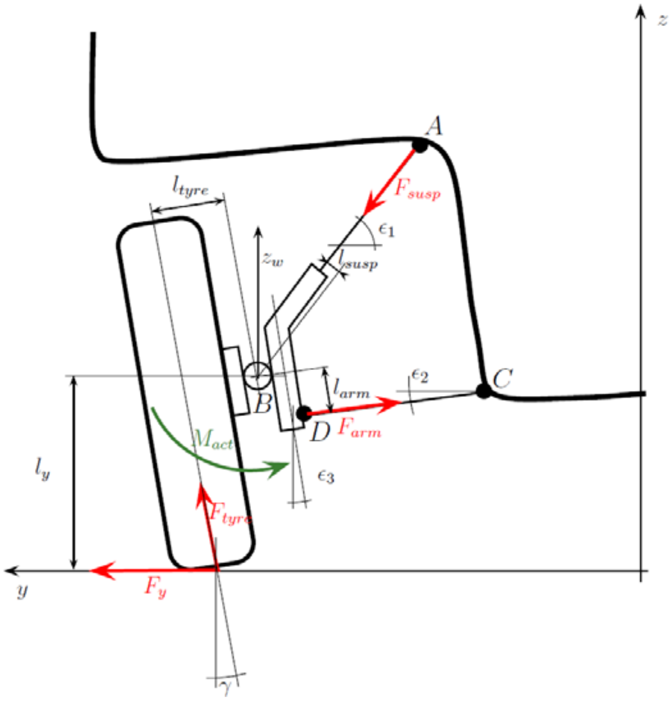


Fig. 1 The scheme of the suspension construction

In the practice ε_3 is constant, thus $\ddot{\varepsilon}_3$= 0. Therefore, F_{arm} is computed as:

$$F_{arm} = \frac{M_{act} - F_{susp} l_{susp}}{l_{arm}} \tag{3}$$

Thus, it is assumed that $\varepsilon_1, \varepsilon_2$ and l_{arm}, l_{susp} can be handled as constant suspension parameters. More detailed descriptions of the nonlinear suspension model can be found in [15], [16].

3. VALIDATION OF THE NONLINEAR MODEL

In this section the validation of the nonlinear suspension model is presented. The nonlinear variable-geometry suspension model is validated through the complex mechanical simulation system Matlab/SimMechanics. The construction of the suspension has been built in SimMechanics, see Fig. 2.

Each component of the suspension has been modelled as a rigid body. The motion of the suspension has been guaranteed by proper joint elements. The modification of the wheel camber angle has been analyzed. Fig. 3 shows a simulation example, in which the suspension nonlinear model and the SimMechanics model are compared. In the simulations the same chirp input signals M_{act} are realized, see Fig. 3(a). The camber angle of the model is presented in Fig. 3(b). It shows that the resulting camber angles are very close to each other in a high operation range.

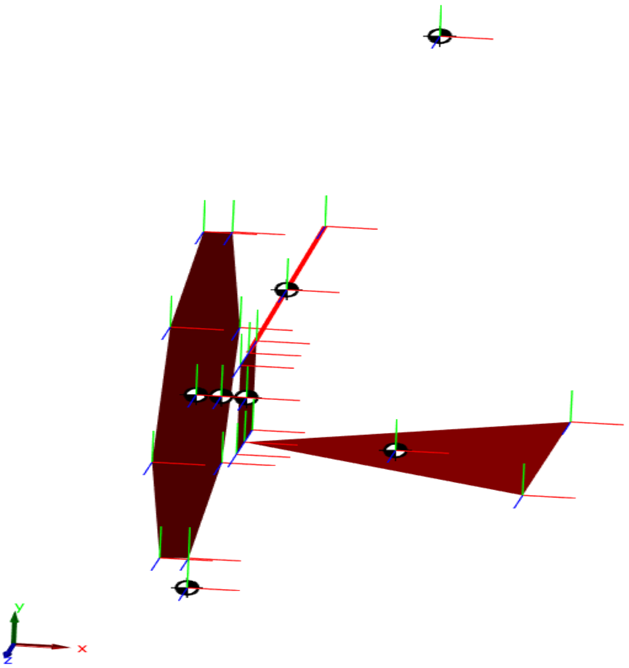
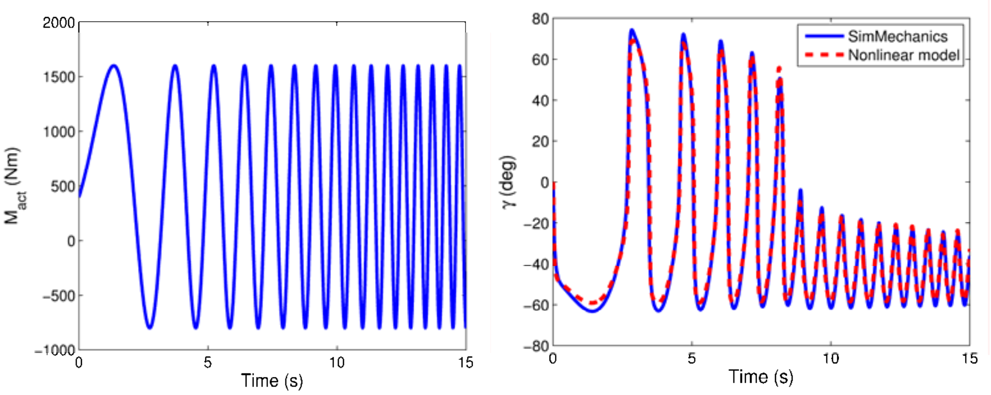


Fig. 2 Suspension model in SimMechanics

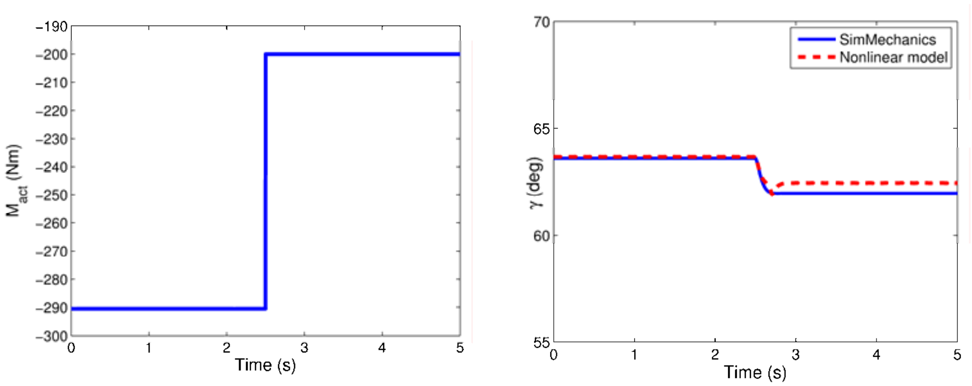
Figure 4 shows another example for the validation of the nonlinear suspension model. The step signal of the control input M_{act} is shown in Fig. 4(a). As an effect of the torque modification, the camber angle also varys, see Fig. 4(b). The results show that the steady-state error of the camber angle is below 2%. It means that the proposed nonlinear suspension model (3) well fits the SimMechanics model.



(a) Control input M_{act} (b) Camber angle (γ)

Fig. 3 Model validation - Chirp signal

Since the presented model formulates the motion of the suspension and the wheel, it can be used for control design purposes. Therefore, the nonlinear model is transformed into a linear control-oriented form.



(a) Control input M_{act} (b) Camber angle (γ)

Fig. 4 Model validation - Step signal

During the linearizing the following assumptions are made.

- In the formulation small wheel tilting angels are considered. As a result $\cos\gamma = 1$ and $\sin\gamma = \gamma$.
- Since γ values are small, the lateral tyre force F_y is approximated in a linear form: $F_y = C\alpha$, where α is the side-slip angle of the tyre. During the wheel tilting motion $\alpha = tan\left(\frac{r_w\dot{\gamma}}{v}\right) \approx \frac{r_w\dot{\gamma}}{v}$ results in the lateral side-slip angle, which is the angle between the longitudinal and the lateral component of the velocity vector. Thus, the resulting lateral tyre force is
$$F_y = C\,\frac{r_w\dot{\gamma}}{v} \tag{4}$$
- The static compressions of the suspension and the tyre are neglected see (1) and (2).

The state-space representation of the variable-geometry suspension model can be obtained from the control-oriented form, see (5).

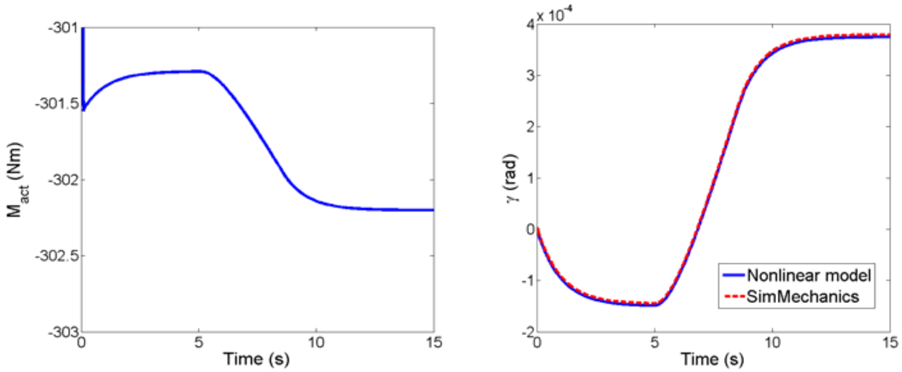
$$\dot{x} = Ax + Bu \tag{5}$$

where the state vector is $x = [\dot{z}_w\ z_w\ \dot{\gamma}\ \gamma]^T$ and the control signal is $u = M_{act}$.

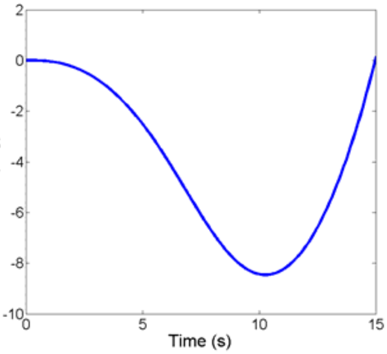
4. SIMULATION EXAMPLE

The aim of the simulations is to show the relationship between the modification of the camber angle and the actual steering angle. The simulations are performed through the complex mechanical simulation system Matlab/SimMechanics.

Fig. 5 (a) shows the intervention on the suspension. The maximum value of the active torque is around $302\ Nm$, which is a reasonable value for the suspension actuator. The camber angles are shown in Fig. 5 (b). It can be seen that the different between the nonlinear model and the SimMechanics model is small. Finally, Fig. 5 (c) presents the resulting steering angle.



(a) Suspension actuation M_{act} (b) Camber angle (γ)



(c) Steering angle (δ)

Fig. 5 Simulation results

5. CONCLUSION

The paper has presented a new independent steering system, which based on the variable-geometry suspension. The nonlinear model of the suspension and its validation has been presented through a complex mechanical simulation system. The main contribution of the paper is to transform the nonlinear model into a control-oriented form. The transformation of the nonlinear equations has been presented. Finally, high-fidelity simulations have demonstrated the operation of the proposed system.

The paper proposed that the variable-geometry suspension can be an alternative way to independent steering control. Since in-wheel electric vehicles are likely to have a significant impact in the future, further research on the variable-geometry suspension control is reasonable.

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